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Received on: 24/05/2019
Accepted on: 22/06/2019
Published online: 25/10/2019

Design and Control of a Full-Scale Quarter Car Test Rig for Semi-Active Suspension System

Abstract- *Passive hydraulic dampers are commonly used in the automotive suspension system. Nevertheless, they are suffering from a significant drawback owing to the changing of its characteristics at high-frequency; as a result, decreasing the ride quality due to the increase of the transmitted force, especially at high frequency excitations. The present work developed a semi-active suspension system to solve this problem with its effect. A Sky-hook control strategy is used to suppress the positional oscillation of the sprung mass in the presence of road irregularities via the use of the electrohydraulic (EH) damper, as an objective. In order to apply the control strategy used herein, a full-scale quarter-car test platform has been designed and constructed to offer increased testing flexibility at a reasonable cost not found commercially. MATLAB Simulink is applied for modeling the semi-active suspension system. The control strategy using a Sky-hook control was used to enhance the comfort due to the simplicity of this method that can easily be implemented in a real-time embedded application. The control strategy is evaluated for its performance under the road bump excitation. The experimental results were compared with the simulated ones for both passive and semi-active suspension systems, the comparison includes time response analysis of body vertical displacement, and vertical displacement of quarter car structure.*

Keywords- *Suspension System; Robust Control, full scale quarter-car, test rig*

How to cite this article: B.R. Abdullah, M.N. Hamzah and A.S. Merza, "Design and Control of a Full-Scale Quarter Car Semi-Active Suspension System," *Engineering and Technology Journal*, Vol. 37, Part A, No. 10, pp. 416-421, 2019.

1. Introduction

The primary function of the suspension system is to isolate the frame of the vehicle from road irregularities. A distinctive car with a suspension system is needed to maintain a regular contact between the road and the tires of the vehicle. The damper is considered as a highly significant element of this system. It decreases the results of an unanticipated bump on the road via reducing the roughness of shock. In the majority of absorbers of shock, the energy of vibration is first changed to heat and then dispelled into the surrounding. In a viscous damper, the energy is transformed into heat by a viscous fluid. In the hydraulic cylinders, the fluid becomes hot [1]. Vibration control techniques are widely categorized into three types of controls, including passive, semi-active, and active controls. A passive system of suspension is a control system considered as an open loop to conduct only a specific state, therefore the characteristics of this kind of suspension are fixed and cannot be modified by any mechanical component, and road irregularities are transformed considerably. Although considering the tire damping has a slight effect on the wheel-hop vibration, the ignorance of damping in tire models compelled the misleading conclusions that at the wheel hop frequency. Moreover an active suspension possesses an enhanced suspension performance by getting a force actuator, which is a control system having a closed-loop [2]. The semi-active suspension system gains popularity in the automotive suspension system due to its higher efficiency and advantageous features over the passive one. Within the systems of semi-active suspension, the properties of damping of a damper are modified to a certain extent. The modified properties in the semi-active dampers are performed through a diversity of technologies, as piezoelectric actuators, electrorheological (ER) and magnetorheological (MR) fluids, and solenoid valves. It has been broadly recognized that a system of semi-active suspension gives higher efficiency than a passive type system [1].
In recent years, the popularity of semi-active suspension attracted much interest to many researchers, e.g., the semi-active system that used the magnetorheological damper (MRD) that has the adjustable controlled force that varies with the current. Other kinds of semi-active control algorithms are, e.g, sky-hook control, state-based

http://dx.doi.org/10.30684/etj.37.10A.6
2412-0758/University of Technology-Iraq, Baghdad, Iraq

control algorithms [3], which successfully applied to vibration control of vehicle suspension systems. Other researchers tackled the nonlinearity of MR damper dynamics by using different control strategies such as neural networks, fuzzy logic, and sliding mode control [4, 5].
The main objective of the present work is to design and manufacture a test rig platform to check the performance of the automotive semi-active suspension system. The control strategy used was a Sky-hook control to suppress the positional oscillation of the sprung mass in the presence of road irregularities via the use of the electrohydraulic (EH) damper.

2. Manufacturing of Test Rig

The purpose of the manufactured test rig is to construct a full-scale platform test rig to study the performance of the independent automotive suspension system. Figure 1 shows a SolidWorks model of the test rig with its primary components. The dimension of the base plate used in this design is 110 cm × 70 cm × 10 cm; the purpose of it is to build a steel base plate for the rig. The base has a trolley guide slot that carries the bump, which allows it to move in a linear motion. The trolley is connected to a pneumatic cylinder and moved by a specific distance along with the piston.

3. Measuring Devices

There are two reasons for using accelerometers. Firstly, an acceleration signal is used as an indicator for ride comfort. Secondly, the car body and wheel unit velocity need to be measured and used in feedback control. Theoretically, a velocity signal could be obtained from an acceleration signal by an analog or digital integrator.

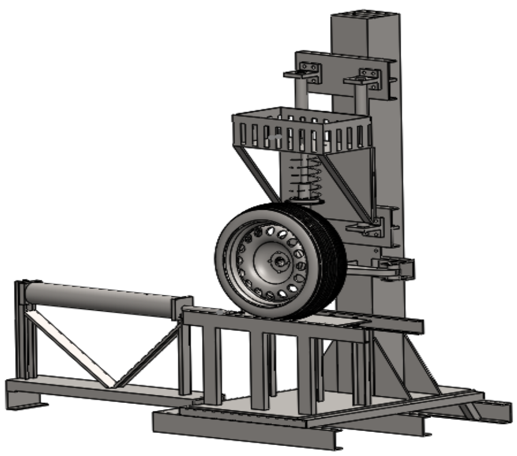

Figure 1: Quarter Car Test Rig

4. Determining Damping Coefficient

The damping coefficient is determined by relating the damper forces characteristics with their compression and extension velocities. However, the essential parameters stated by these characteristics is by correlating it with the ride and handling quality, which represents an overall mean damping coefficient [6]. As the electrohydraulic (EH) manufacturer data was not available, a test rig was constructed to determine the damping coefficient experimentally. This test rig is shown in Figure 2, and Figure 3 shows the experimental results.

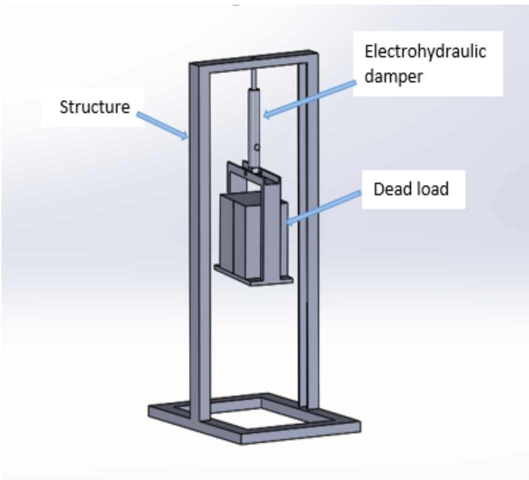

Figure 2: Determining the Damping Coefficient Experimentally

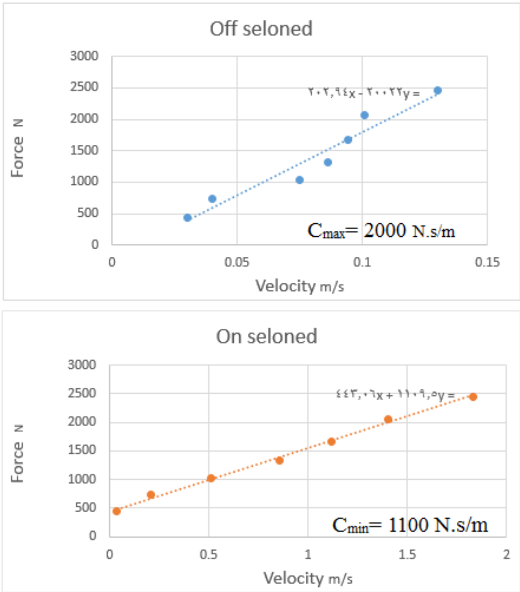

Figure 3: Finding the Damping Coefficient Experimentally by Using the Test Rig.

5. Quarter Car Mathematical Model

In this paper, the theoretical analysis is based on a 2-DOF quarter car model; the model considered linear behavior within the suspension spring and

damper forces that have a vital influence on the responses of the suspension when excited by completely different road inputs [7]. Consider a simple model of quarter car, shown in Figure 4, made up of a sprung mass (m_s) and an unsprung mass (m_{us}). Spring with the stiffness coefficient k_s and a semi-active damper connected both masses. A spring modeles the wheel tire with the stiffness coefficient k_t. In this model, z_1 and z_2 are the vertical position of m_{us} and m_s, respectively.
In Figure 4, the free body diagram leads to the following mathematical equations of motion:

$$m_s\ddot{z}_2 + c(\dot{z}_2 - \dot{z}_1) + k_s(z_2 - z_1) + f_d = 0 \quad (1)$$

$$m_{us}\ddot{z}_1 - c(\dot{z}_2 - \dot{z}_1) + z_1(k_t + k_s) - k_t z_\circ - k_s z_2 - f_d = 0 \quad (2)$$

This can be used to get a state-space representation [8] described in the following section.

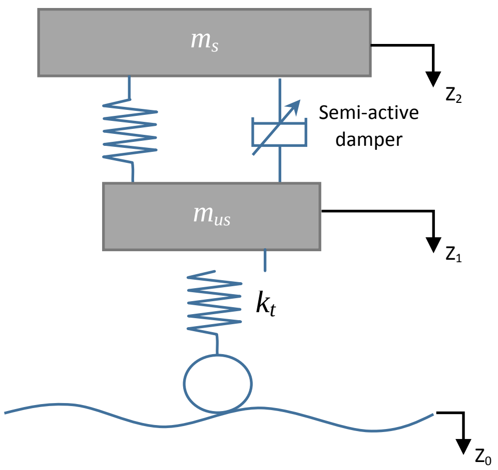

Figure 4: Model of quarter car with a semi-active suspension

Table 1: Test results of deep beam specimens

Parameter	Value
Sprung mass (m_s)	250 kg
Unsprung mass (m_u)	41 kg
Suspension stiffness (k_s)	37745.6 N/m
Tire stiffness (k_t)	190000 N/m
Suspension damping c_1	2000 N.s/m
Suspension damping c_2	1100 N.s/m

6. State Space Model

$X_1=Z_1, X_2=Z_2, X_3=Z_1', X_4=Z_2'$, then the system state equation can be expressed as:

$$X'=AX+BU \quad (3)$$

$$X'=[X_1, X_2, X_3, X_4];\quad U=[Z_o, F_d] \quad (4)$$

Of which

$$X'_1= 0\times X_1 + 0\times X_2 + X_3 + 0\times X_4 \quad (5)$$

$$X'_2= 0\times X_1 + 0\times X_2 + 0\times X_3 + X_4 \quad (6)$$

$$X'_3=\frac{-kt+ks}{mus}\times X_1 + \frac{ks}{mus}\times X_2 - \frac{c}{mus}\times X_3 + \frac{c}{mus}\times X_4 \quad (7)$$

$$X'_4=\frac{ks}{ms}\times X_1 - \frac{ks}{ms}\times X_2 + \frac{c}{ms}\times X_3 - \frac{c}{ms}\times X_4 \quad (8)$$

State space equation can be written as form,

$$\begin{bmatrix} x'_1 \\ x'_2 \\ x'_3 \\ x'_4 \end{bmatrix} = \begin{bmatrix} 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \\ \frac{-kt+ks}{mus} & \frac{ks}{mus} & \frac{-c}{mus} & \frac{c}{mus} \\ \frac{ks}{ms} & \frac{-ks}{ms} & \frac{c}{ms} & \frac{-c}{ms} \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \end{bmatrix} + \begin{bmatrix} 0 & 0 \\ 0 & 0 \\ \frac{kt}{mus} & \frac{1}{mus} \\ 0 & \frac{-1}{ms} \end{bmatrix} U \quad (9)$$

where,

$$A = \begin{bmatrix} 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \\ \frac{-kt+ks}{mus} & \frac{ks}{mus} & \frac{-c}{mus} & \frac{c}{mus} \\ \frac{ks}{ms} & \frac{-ks}{ms} & \frac{c}{ms} & \frac{-c}{ms} \end{bmatrix}, B = \begin{bmatrix} 0 & 0 \\ 0 & 0 \\ \frac{kt}{mus} & \frac{1}{mus} \\ 0 & \frac{-1}{ms} \end{bmatrix} \quad (10)$$

System output equation is:

$$Y=CX+DU \quad (11)$$

$$C=\begin{bmatrix} -ks & 0 & 0 & 0 \\ \frac{ks}{ms} & \frac{-ks}{ms} & \frac{c}{ms} & \frac{-c}{ms} \\ 1 & -1 & 0 & 0 \end{bmatrix}, D=\begin{bmatrix} kt & 0 \\ 0 & \frac{-1}{ms} \\ 0 & 0 \end{bmatrix} \quad (12)$$

7. Control Methods

Better controller design significantly improves the vehicle ride and handling. In this paper, the sky-hook control strategy is used to adjust the damping force so that improving the suspension dynamics. The Sky-hook control is the most widely used controlling algorithm insensible and covers some types of scientific domains [9].

8. Sky-hook Control Method

The sky-hook control is one of the most effective control algorithms due to its simplicity. The sky-hook can reduce the resonant peak of the vehicle sprung mass significantly; as a result improved ride quality, this is performed by adjusting the skyhook damping coefficient as the vehicle body velocity, and other conditions are changing, the adjustable damping ranges between hard and soft envelopes [10,11]. The approximation to an ideal sky-hook can be expressed by the following control logic [12-14].

$$f_{sky} = c_{sky}\,(\dot{x}_b - \dot{x}_w) \quad (13)$$

Where, f_{sky} is the force applied to the sprung mass. As the force in the proper direction can be generated, the skyhook suspension requires matching the following:

$$c = \frac{c_{sky} \times \dot{x}_b}{(\dot{x}_b - \dot{x}_w)} \quad (14)$$

$$c = \begin{cases} c_{max} \; if \; \dot{x}_b \times (\dot{x}_b - \dot{x}_w) \geq 0 \\ c_{min} \; else \; \dot{x}_b \times (\dot{x}_b - \dot{x}_w) < 0 \end{cases} \quad (15)$$

9. Numerical Simulation

The effectiveness of the suggested control strategy is studied using MATLAB; the purpose is to explore the effect of different system parameters on the sky-hook control law. The main objective is to increase or to evaluate the passenger ride comfort during the traveling of the vehicle under bump type road excitation. Simulation by MATLAB m-file was achieved in the time domain for both passive and semi-active (controlled) of the quarter car suspension systems using On/Off Sky-hook controllers. The simulation run time was set to 10 s, and the vehicle traveling speed is considered to be 20 km/h, as shown in Figure 5.

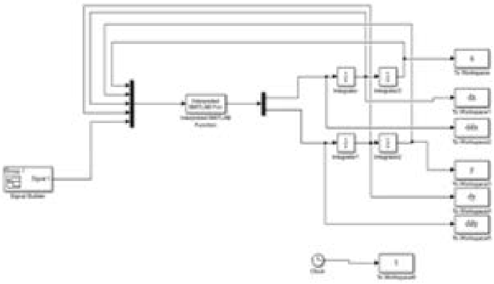

Figure 5: Semi-active suspension simulation block diagram

10. Results

The numerical simulation exhibited the effectiveness of the Sky-hook control technique in controlling the suspension while driving on rugged terrain. The results shown in figure 6 and Figure 7 were obtained. The starting time for the tire to bump is (4.5) seconds. The maximum overshoot is 0.024/m at a time (4.8) seconds, and the maximum undershoot is -0.006m at a time (5.3) seconds. The time of the stability of the system at (7) seconds. And Figure 7 shows the sprung mass acceleration. The maximum overshoot is (5.85) m/s2 at a time (4.5) seconds, and the maximum undershoots -5.85 m/s2 at a time 5.1 seconds.

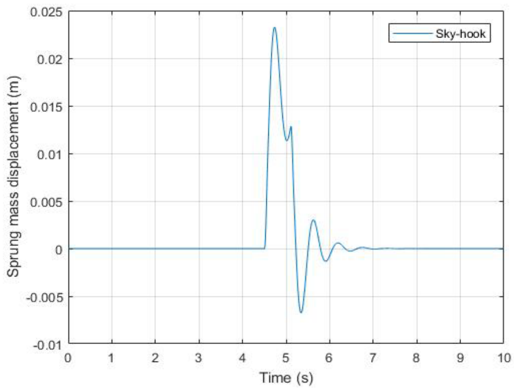

Figure 6: Displacement for sprung mass.

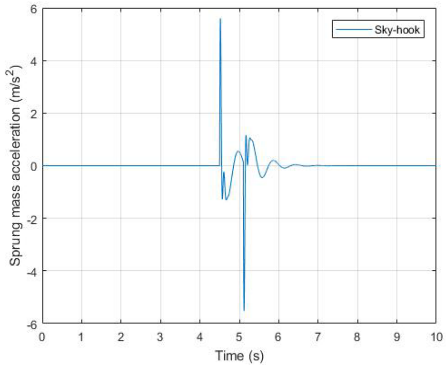

Figure 7: Acceleration for sprung mass.

Figures (8 -9) show the experimental results of the semi-active suspension using the control technique Sky-hook. Figure 8 illustrates the sprung mass displacement representing the body of the car, where the maximum overshoot is 0.023/m at the time of 4.6 seconds, and the maximum undershoot is -0.008m at the time of 5 seconds. Because of the continuous Sky-hook on/off control method note the descent of the line is a breaker unlike the semi-active system without control. Also, the system stability time is 5.3 seconds less than the time of the semi-active system without control. Figure 9 demonstrates, the acceleration amplitude, the peak value is reduced to 39% percent and also the settling time is reduced, giving better ride performance.

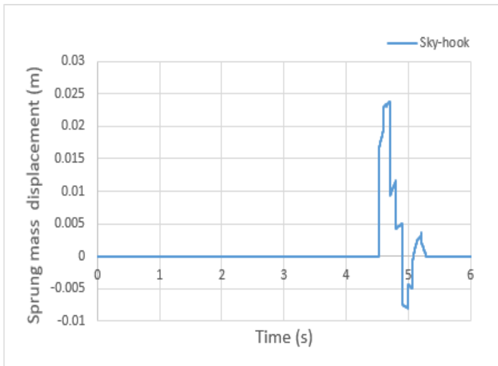

Figure 8: Displacement for sprung mass.

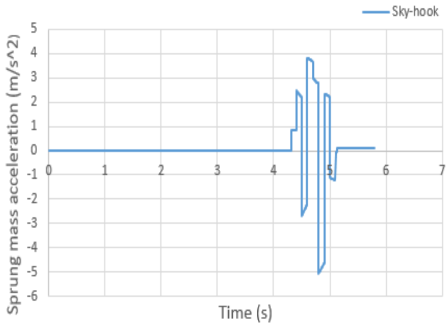

Figure 9: acceleration for sprung mass.

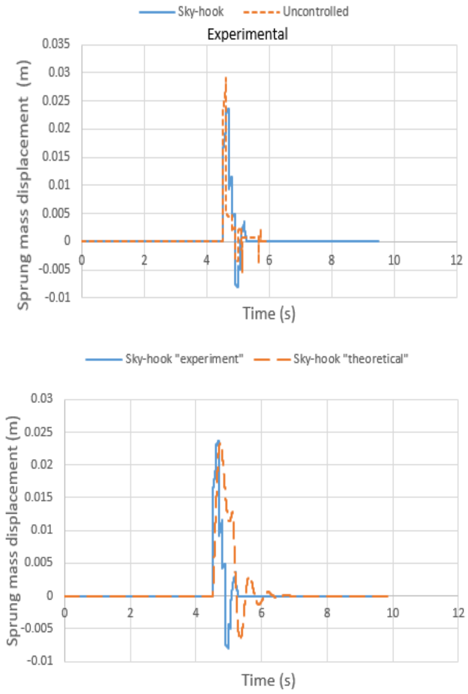

Figure 10: Comparison between experiment and theoretical results.

Figure 10 shows the comparison between a semi-active controlled suspension system and the other semi-active suspension system without control experimentally. It was observed through the form that the control technique used reduced the maximum overshoot to 20% and also reduced the stability of the semi-active suspension to 27%.

11. Conclusion

The conventional sky-hook offers the best performance at low frequencies close to the first mode of resonance and reasonably good performance at higher frequencies as well. From the result of the simulation, the semi-active suspension reduced the peak amplitude and settling time to 53% and 67%, respectively than the passive suspension. Also, the performance of the semi-active suspension under the control effect gives better performance than the semi-active uncontrolled system.

Acknowledgement

The authors are grateful to the Department of Mechanical Engineering، The University of Technology for help and providing testing facilities.

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